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D. N. Olaru / P. Lorenz / Gh. Prisacaru

Friction Coefficient in a Micro Linear Rolling System for MEMS Applications

ABSTRACT

The authors propose an analytical model to evaluate the friction coefficient in a micro linear ball bearing. The proposed model includes both the friction losses developed in a macro rolling tribosystems and the adhesion and capillary effects specified to microsystems. The numerical values obtained by this model are in correlation with the experimental results obtained in [1]. Also, the authors determined the rolling friction coefficient in a micro rolling tribosystem based on the free oscillations of a ball on a spherical surface. The experimental results confirm the high influence of the capillary forces on the rolling friction coefficient.

1. INTRODUCTION

The nanotechnology, described often as the technology of the future focused the researches and development activities in the last two decades. A lot of miniaturized devices, known as MEMS (micro electromechanical systems) was realised both for research in the laboratories and for various applications in the automotive industry, medical instrumentation, informatics technology as: micro sensors, linear and rotary micro actuators, micro motors, micro pumps, micro gear transmissions, micro grippers. The MEMS devices include mobile components with dimensions having the order of magnitude of about 10^{-6} m. When the dimension of a machine component decreases from millimetres to microns, the area decreases by a factor of 10^6 and the volume (mass) decreases by a factor of 10^9 . In the motion of such components the resistive forces as friction, viscous drags or surface tensions that are proportional to the area increase by a factor of 10^3 or more that the inertial forces that are proportional with the mass of the elements. As a result of small dimensions friction on the moving surfaces in the MEMS becomes critical and is one of the fundamental limitation in the design and the fabrication of reliable MEMS. Most of the MEMS devices have contact-type bearings with sliding friction, when the friction coefficient can be usually higher than $0.5 \dots 1$ and the stick-slip instabilities can appear in the low speed conditions. Also, the wear limits the performances of such devices. Non contact-type bearings with more complicated support mechanisms like electrostatic or pressurized air have much less friction and non wear compared to contact-type bearings but are more complex systems. The use of the micro linear or rotating ball bearings in the MEMS applications implies the simplify in construction, low level of friction and

high stability, so that the micro ball bearings seems promising for future MEMS applications. The first micro linear ball bearing structure was studied experimentally by Ghodssi et al. in 1993 and the static coefficient of friction was reported as low as 0.056 [1]. Recent researches realised by Ta-Wei Lin et al. [1] with sophisticated equipment determined the static and dynamic coefficient of friction for a micro ball linear system. By determining the acceleration in a linear micro ball structure, the authors established that the dynamic coefficient of friction is between 0.007 to 0.045 if is not interaction between balls. If the balls interacts the dynamic coefficient of friction can increases up to 0.6.

In this paper the authors propose an analytical model to evaluate the friction coefficient in a micro linear ball bearing. The proposed model includes both the friction losses developed in the macro rolling tribosystems [2] and the adhesion and capillary effects specified to microsystems. The numerical values of the coefficient of friction obtained by the developed model are in correlation with the experimental results obtained in [1].

2. FRICTION IN THE MICRO ROLLING TRIBOSYSTEMS

In macrotribology the processes are developed in components with relatively large mass under heavily loaded conditions. In these conditions, wear is inevitable and the bulk properties of mating components dominate the tribological performance. In micro/nanotribology, the tribological processes are developed in systems with relatively small mass under lightly loaded conditions. In this situation, negligible wear occurs and the surface properties dominate the tribological performance.

The components used in micro- and nanostructures are very light (on the order of a few micrograms) and operate under very light loads (on the order of a few micrograms to a few milligrams). As a result, friction and wear (on a nanoscale) of lightly loaded micro/nanocomponents are highly dependent on the surface interactions (few atomic layers). These structures are generally lubricated with molecularly thin films. In the micro and nanotribosystems a lot of interfacial forces as adhesion, van der Waals, electrostatic, capillary forces can be important and have an important contribution on the friction losses. Must be observed that in micro and nanotribosystems the friction forces and moments can stopped the normal function.

Adhesion between two solid surfaces based on the thermodynamic interfacial free energy can develop attraction forces of 200 - 300 μ N or more [3,4,5]. For a ball on a flat surface the adhesive force F_a is given by relation [3]:

$$F_a = 3 \cdot \pi \cdot R \cdot \gamma \quad (1)$$

where R is the radius of the ball and γ is the interfacial energy (J/m^2).

For two balls in contact with radius R_1 and R_2 , the adhesive force is given by relation:

$$F_a = 3 \cdot \pi \cdot \gamma \cdot [1/R_1 + 1/R_2]^{-1} \quad (2)$$

As a result of adhesion, a micro-rolling tribosystem can be loaded supplementary with normal force and the contact area A_c increases. According to the Johnson- Kendall-Robert (JKR) model [3], the contact area between a ball and a plane with including the adhesion effect is given by relation:

$$A_c = \pi \cdot \left[\frac{R}{E^*} \cdot \left(Q_0 + 6 \cdot \pi \cdot \gamma \cdot R + \sqrt{12 \cdot \pi \cdot \gamma \cdot R \cdot Q_0 + (6 \cdot \pi \cdot \gamma \cdot R)^2} \right) \right]^{2/3} \quad (3)$$

where Q_0 is normal force applied to the ball. Even in absence of an applied a normal load, the ball stick to the surface.

E^* is the equivalent elastic modulus for the two solids in contact:

$$E^* = \frac{4}{3} \cdot \left[\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right]^{-1} \quad (4)$$

where ν and E are the Poisson ratio and Young's modulus, respectively.

The capillary forces are presents as a result of the condensed water from atmosphere on the solids. The most of the solids are hydrophilic surfaces and the atmospheric water cover these surfaces with molecular layers. In the contact zone between the two solids, by the capillary effect the adhered water lead to increase of normal force. A lot of experiences evidenced the influence of the pressure, temperature and humidity of air on the thickness of the condensed water films. So, Opitz, A. et al [6] measured for hydrophilic silicon the water layers between 0.7 nm to 2.6 nm, depending of the pressure. Other experiments presented in [3-5] indicated the water layers of 10 to 50 nm, for various materials. The capillary effects are dominant for the water layers more than $(0.7 \dots 1) \text{ nm}$ [3,6]. For a ball on a flat surface, as in figure 1, the ball is attracted on the flat surfaces by a capillary force F_c given by relation [3]:

$$F_c = \Delta p \cdot A_l \quad (5)$$

where Δp is the Laplace pressure and A_l is the area that is covered by the liquid.

The Laplace pressure is depending of the curvature radius of the meniscus realised by water in the interstice between the two solids [3]:

$$\Delta p = \gamma \cdot (1/r_1 + 1/r_2) \quad (6)$$

where γ is interfacial energy (N/m), r_1 is the meniscus radius in the normal plane and r_2 is the meniscus radius in the tangential plane.

The area covered by liquid can be obtained by relation:

$$A_l = \pi \cdot r_2^2 \quad (7)$$

If h_l is the thickness of the water film between ball and plane at the level of meniscus, both the two radius r_1 and r_2 can be expressed as function of h_l with sufficient approximation.

For very small values of h_l the capillary force between a ball and a plane is given by relation:

$$F_c = 2 \cdot \pi \cdot R \cdot \gamma \cdot \left(2 + \sqrt{\frac{h_l}{2 \cdot R}} \right) \quad (8)$$

For $h_l \ll R$, the capillary force F_c is not depending of the meniscus radius and relation (21) can be replaced with the well known relation [3]:

$$F_c = 4 \cdot \pi \cdot R \cdot \gamma \quad (9)$$

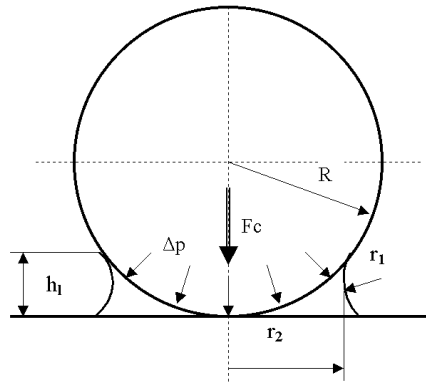


Figure 1: Ball-plane interaction as result of capillary force

For a ball-ball contact the radius R in relation (9) will be changed by the equivalent radius R^* given by relation:

$$R^* = \left[1/R_1 + 1/R_2 \right]^{-1} \quad (10)$$

Sliding friction results in micro and nanotribosystems

In the last 15 years a lot of researches was realised to determine the realistic friction forces in sliding microsystems. The most important step was the development of the Atomic Force Microscope (AFM) technique in 1985 by Binnig et al. [3]. With the AFM technique ultra small forces (less than $1 \mu\text{N}$), both normal and frictional was determined. A lot of new microtester was developed in the last years. Important results regarding friction was obtained in the Institut für Physik, TU Ilmenau and IAVF Antriebstechnik AG, Karlsruhe, Germany [3-5] using the sliding ball – plane tribosystem and materials based on silicium.

Some important effects regarding the sliding friction in microtribosystem can be observed:

- The water from the atmosphere adhere on the surfaces in contact and influences the friction forces.
- For high water layers (more than 10nm) friction is dominated by capillary. The capillary bridge increases the normal force and imposes resistance against shear. High values for the friction coefficient was experimentally obtained ($\mu = 1 \dots 4$ and more).

- For very low water films (less than 0.2 nm) the low values of the coefficient of friction was determined ($\mu = 0.3 \dots 0.4$).

3. AN ANALYTICAL MODEL TO EVALUATE FRICTION LOSSES IN A MICRO LINEAR BALL BEARING

In the figure 2 is presented the geometry of a micro linear ball bearing. The slider is in relative linear motion from the stator with the linear speed v and the ball has a angular velocity ω_b .

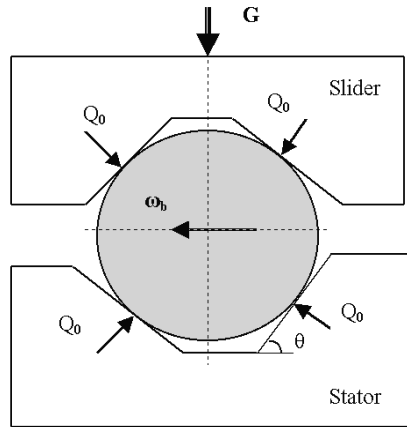


Figure 2: The geometry of the micro - linear ball bearing

Under the external load G all the four contacts of the ball with slider and stator are loaded with a normal force Q_0 given by relation:

$$Q_0 = \frac{G}{2 \cdot z \cdot \cos \theta} \quad (11)$$

According to the JRK adhesion model the normal force for every ball – race contact Q will be higher than Q_0 and we propose to be computed by following relation:

$$Q = \left(Q_0 + 6 \cdot \pi \cdot \gamma \cdot R + \sqrt{12 \cdot \pi \cdot \gamma \cdot R \cdot Q_0 + (6 \cdot \pi \cdot \gamma \cdot R)^2} \right) \quad (12)$$

Considering thin water layers on the surfaces of balls and races as lubricant, in all four contacts are developed following forces [2]: hydrodynamic rolling forces FR , pressure forces FP , sliding contact forces FS .

The directions of these forces acting on the ball are presented in figure 3. Note that the index 1 refer to slider and index 2 refer to stator. The hydrodynamic forces $FR1$ and $FR2$ are computed with relation:

$$FR = 2.86 \cdot E \cdot R_x^2 \cdot k^{0.348} \cdot G^{0.022} \cdot U^{0.66} \cdot W^{0.47} \quad (13)$$

where E is the equivalent Young's modulus of the materials in contact, R_x is the equivalent radius in the rolling direction, k is the radii ratio.

G is the dimensionless material parameter,

$$G = E \cdot \alpha_p \quad (14)$$

where α_p is the piesoviscozitat coefficient.

U is the dimensionless speed parameter,

$$U = \frac{\eta_0 \cdot v}{E \cdot R_x} \quad (15)$$

where η_0 is the viscosity of the lubricant at the atmospheric pressure and at the contact temperature, v is the tangential speed in the ball-races, in the rolling direction.

W is the dimensionless load parameter definite by relation:

$$W = \frac{Q}{E \cdot R_x^2} \quad (16)$$

where Q is normal load in the ball-race contact.

FP is pressure forces due to the horizontal component of the lubricant pressure in the rolling direction. For a ball – ring contact, the pressure force acting on the centre of the ball can be expressed as a function of hydrodynamic rolling force FR:

$$FPb = 2 \cdot FR \cdot \frac{R_R}{R_R + R_b} \quad (17)$$

where R_b is the ball radius and R_R is the ring radius.

The friction forces FS on the two contacts are the sliding traction forces due to local micro slip occurring in the contact, and can be calculated explicitly as the integral of the shear stress τ over the contact area:

$$FS = \int \tau \cdot dA \quad (18)$$

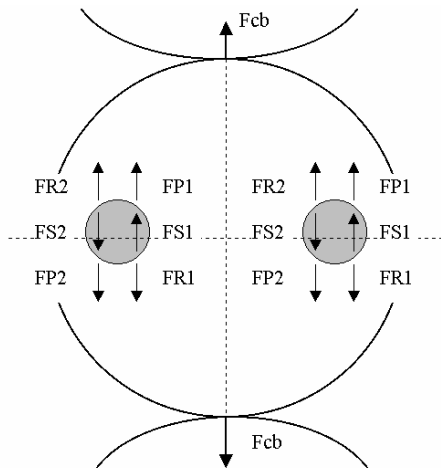


Figure 3: Forces acting on a ball in rolling direction

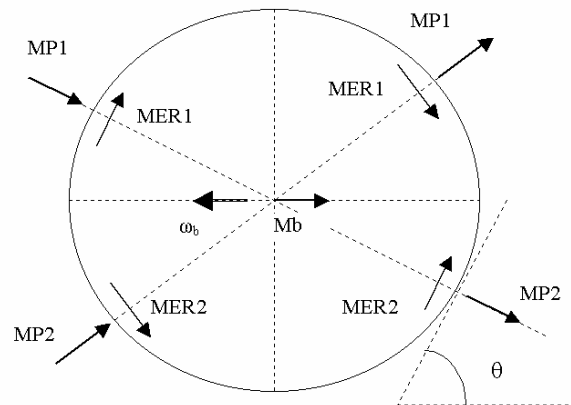


Figure 4: The moments acting on a ball

The capillary force between balls, Fcb is computed with relations (9). The friction forces FS1 and

FS2 can not be easily determined directly by relation (18). Shear stress on the ball race contacts τ is depending of microslidings on contact surface, rheology of thin water layers, contact pressure.

Based on the equilibrium of the forces and moments acting on the ball, the friction forces FS1 and FS2 can be determined as result of all the other forces and moments acting on the ball.

The moments acting on the ball are presented in figure 4. In each ball-race contact, two resistance moments are developed: the elastic resistance moments MER1 and MER2 and the pivoting moments MP1, MP2.

$$MER = 7.48 \cdot 10^{-7} \cdot \left(\frac{d}{2}\right)^{0.33} \cdot Q^{1.33} \cdot \left[1 - 3.519 \cdot 10^{-3} \cdot (k-1)^{0.806}\right] \quad (19)$$

where d is the ball diameter.

Considering a constant friction coefficient for the sliding motion in a ball-race contact μ_s , the pivoting moment MP can be evaluated by relation:

$$MP = \frac{3}{8} \cdot \mu_s \cdot Q \cdot a \quad (20)$$

The semi major contact ellipse axis a is computed considering both normal load Q_0 and adhesion, according to JRK model. So, from equation (3) results:

$$a = \left[\frac{R}{E^*} \cdot \left(Q_0 + 6 \cdot \pi \cdot \gamma \cdot R + \sqrt{12 \cdot \pi \cdot \gamma \cdot R \cdot Q_0 + (6 \cdot \pi \cdot \gamma \cdot R)^2} \right) \right]^{1/3} \quad (21)$$

As a result of contacts between balls, for each ball acts a resistance moment M_b evaluated by relation:

$$M_b = \mu_b \cdot F_{cb} \cdot d \quad (22)$$

where μ_b is the friction coefficient between two balls and F_{cb} is the capillary force determined with relations (9)

To determine the friction forces FS1 and FS2, following two equations of equilibrium are written:

- the equilibrium of the forces acting of the ball in the rolling direction,

$$FS1 - FS2 - (FR1 - FR2) + (FP1 - FP2) = 0 \quad (23)$$

- the equilibrium of the moments acting on the ball,

$$\left[(MP1 + MP2) \cdot \tan(\theta) + (MER1 + MER2) + \frac{M_b}{\cos(\theta)} \right] = \frac{d}{2} \cdot (FS1 + FS2 - FR1 - FR2) \quad (24)$$

Solving the equations (23) and (24) it can be obtained relations for the forces FS1 and FS2:

$$FS1 = \frac{1}{d} \cdot \left[(MP1 + MP2) \cdot \tan(\theta) + (MER1 + MER2) + \frac{M_b}{\cos(\theta)} \right] + 2FR2 - FR1 \quad (25)$$

$$FS2 = \frac{1}{d} \cdot \left[(MP1 + MP2) \cdot \tan(\theta) + (MER1 + MER2) + \frac{Mb}{\cos(\theta)} \right] + 2FR1 - FR2 \quad (26)$$

For the same geometry and material for slider and stator the forces FR and the moments MP and MER are the same values both for contact (1) and contact (2). Results the same values for FS1 and FS2 given by relation:

$$FS1 = FS2 = FS = \frac{1}{d} \cdot \left[2MP \cdot \tan(\theta) + 2MER + \frac{Mb}{\cos(\theta)} \right] + FR \quad (27)$$

For every ball-race contact, the ball acts on the race, according to figure 1 with the total tangential force in rolling direction $FS + FR - FP_r$,

Considering that every ball acts on the slider or stator in two contact points and note that the pressure force acting on a linear race is null ($FP_r = 0$) it can be obtained the total tangential resistance force given by a ball in rolling motion:

$$F_{ball-race} = \frac{1}{d} \cdot \left[4MP \cdot \tan(\theta) + 4MER + \frac{2Mb}{\cos(\theta)} \right] + 4FR \quad (28)$$

For a system with z balls on the slider or on the stator acts a total tangential force F_{total} given by relation:

$$F_{total} = z \cdot F_{ball-race} \quad (29)$$

Determination of the coefficient of friction

A global friction coefficient can be obtained by dividing the total tangential resistance of the slider by the normal load G:

$$\mu_{global} = \frac{z \cdot F_{ball-race}}{G} \quad (30)$$

Also, a local contact friction coefficient can be obtained by dividing the total tangential resistance for only a ball-race contact by the normal load applied to contact, Q_0 :

$$\mu_{contact} = \frac{F_{ball-race}}{2Q_0} \quad (31)$$

Numerical results

The numerical results are performed for the condition used in the paper [1]:

- stainless steel balls with diameter $d = 285 \mu\text{m}$;
- silicon slider and stator with two V – grooves realised at an angle $\theta = 54.7^\circ$;
- temperature of 27 degree and a relative humidity of 40%RH;
- relative speed between the slider and stator was between zero to 150 mm/s;

The number of the balls and the load G was considered in two variants:

- Variant A with 4 balls without contact between balls and a mass of the slider $G = 0.9$ grams;
- Variant B with 18 balls on the two V – grooves (9 balls for each V-groove) with the balls in contact one each other and the mass of slider $G = 0.4$ grams

The elastically properties of the ball and races was: $E_{steel} = 2.1 \cdot 10^{11} Pa$, $E_{silicon} = 1.5 \cdot 10^{11} Pa$, $\nu_{steel} = 0.3$, $\nu_{silicon} = 0.3$.

The viscosity of water $\eta = 0.001 Pa \cdot s$ and the piesoviscosit coefficient $\alpha_p = 10^{-8} Pa^{-1}$.

The selection of the friction coefficient values for the pivoting moment between ball ad races μ_s and for the ball-ball contact moment μ_b was made according to the friction measurements realized with the various devices at the micro scale.

Friction coefficient for a system with 4 balls without contacts between balls

The contact ball –race friction coefficient was computed for two values of the friction coefficient in pivoting motion, so for $\mu_s = 0.5$ and for $\mu_s = 1$.

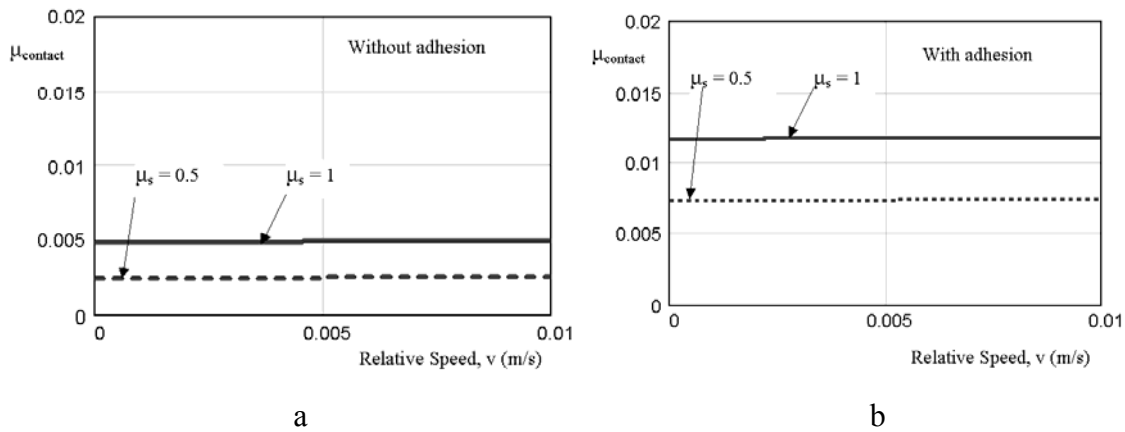


Figure: 5 Contact ball-race friction coefficient determined by analytical model: without ball-race adhesion effect a) and with ball-race adhesion effect b)

If it was neglected the adhesion between ball and race, was obtained values for the contact friction coefficient $\mu_{contact}$ between 0.002 and 0.005, as in figure 5-a. Including the adhesion effects was obtained increasing of the contact friction coefficient with values between 0.007 and 0.012, as in figure 5-b.

Following remarks can be made:

1. According to our analytical model it can be obtained accepted values for the contact ball-race friction coefficient with a magnitude between 0.002 to 0.012, for a micro linear ball bearing. Comparison with experimental values obtained by [1] in the similar conditions evidenced a good correlations between the average values obtained by experiment ($\mu_{contact} = 0.007$) and the computed

values by including adhesion effect ($\mu_{contact} = 0.007...0.012$).

2. No important variation of the friction coefficient with relative speed was obtained by analytical model. It can be explained by the small influence of the hydrodynamic effect given by the forces FR (less of 5%).

3. The most important losses (about 60%) are given by the pivoting motion as a result of geometrical constriction. So, if $\theta \rightarrow 0$, the contact friction coefficient $\mu_{contact} \rightarrow 0.004$.

The variation of the global friction coefficient with normal ball-race force was evaluated analytically and is presented in the figure 6.

It can be observed that important increasing of the global friction coefficient can be obtained by considering higher values for the friction coefficient in pivoting motion ($\mu_s = 1$), from about 0.01 to 0.03. For the same load conditions experimental values of the coefficient of friction obtained by [1] was between 0.007 and 0.045.

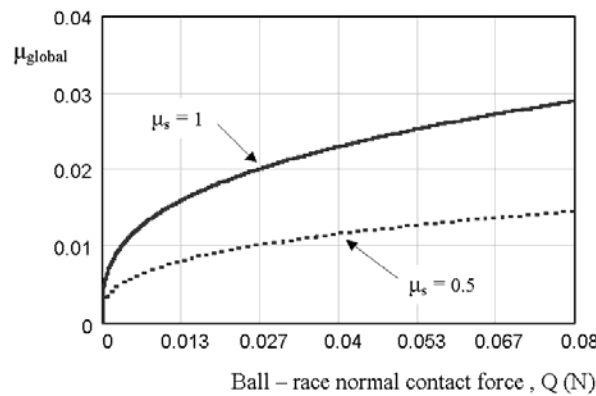


Figure: 6 Variation of the global friction coefficient with ball – race normal force

It can be concluded that the proposed model applied for no contacting ball- ball micro linear ball bearing leads to the values of the friction coefficient compared with experimental values obtained in [1].

Friction coefficient for a system with 18 balls with contacts between balls

The global friction coefficient was computed in the variant B for 18 balls and the mass of slider $G = 0.4$ grams. The friction coefficient for ball-race pivoting motion was imposed $\mu_s = 1$ and two values for the friction coefficient between two balls in contact was imposed :

$\mu_b = 0.5$ and $\mu_b = 1$. The results are presented in figure 7.

The obtained values are between 0.25 to 0.55 for the condition imposed. In the similar conditions, by the experiments realized in [1] it was obtained variation of the friction coefficient between 0.2 and 0.6.

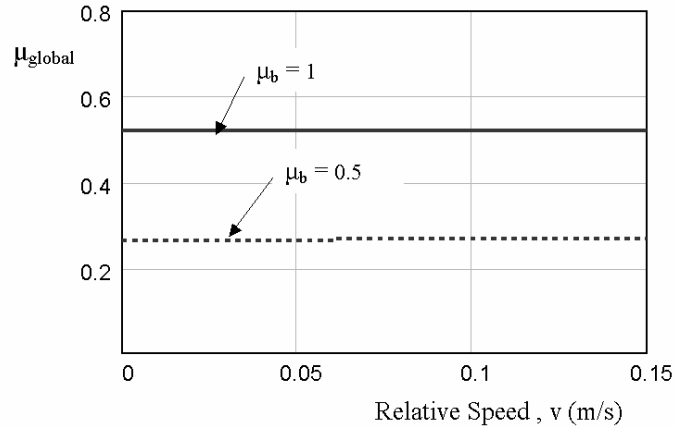


Figure: 7 The global friction coefficient determined by analytical model for 18 balls

It can be observed that the contacts between balls dominates the losses for the given conditions. The influence of the pivoting friction is more smaller that the friction between balls. The model included only the capillary effects between the balls in contact.

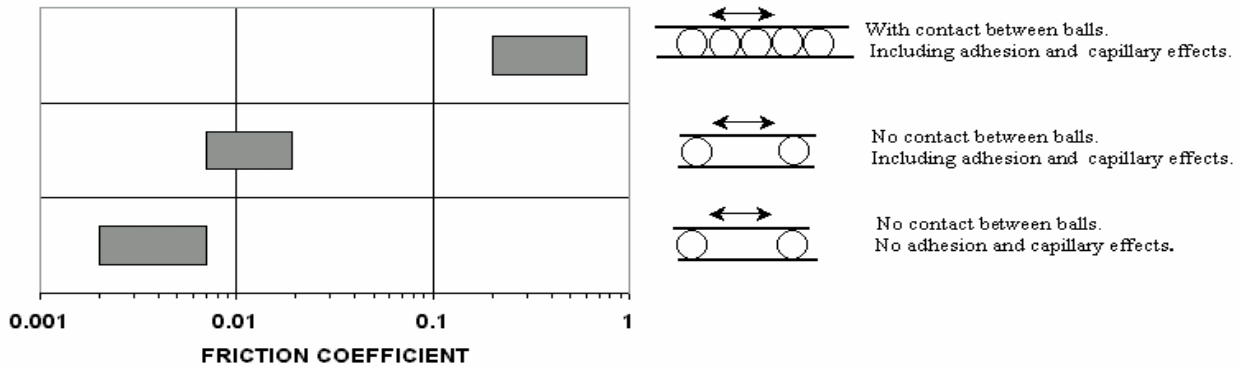


Figure 8 Friction coefficient in a micro linear rolling system

In figure 8 are presented the most important results of friction coefficient in a micro linear rolling system. So, if withouth adhesion and capillary effects the friction coefficient ia a micro linear rolling system has values between 0.002 to 0.008 (similar with a macro linear system). Adhesion and capillary effects esspecialy lead to an important increasing of the friction coefficient with about two order of magnitude.

4. EXPERIMENTAL DETERMINATION OF THE FRICTION COEFFICIENT IN A MICRO ROLLING TRIBOSYSTEM

To determine experimental the friction coefficient in a micro rolling tribosystem authors developed a methodology based on the free oscillation of a ball on a spherical surface only on

the influence of the gravity. In figure 9 is presented the principle of the methodology.

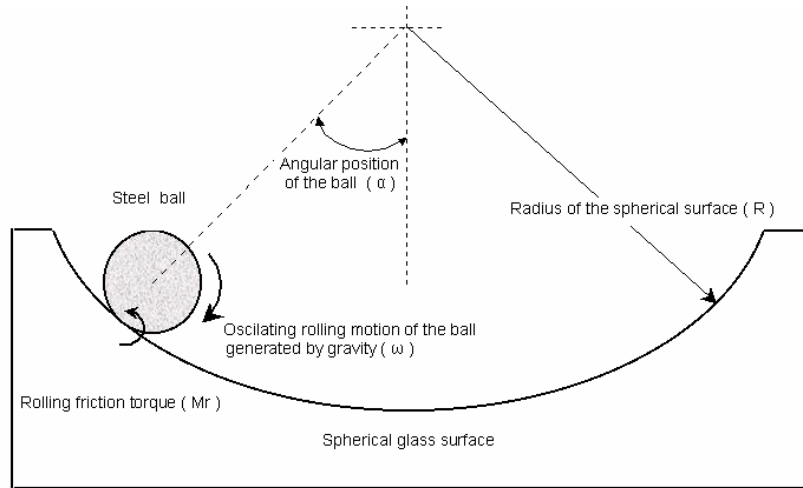


Figure 9 Oscillation of a ball over a spherical surface

When the ball rolling on a spherical surface from a high level to a low level following differential equation can be written:

$$\frac{d^2x}{dt^2} - \frac{5}{7} \cdot g \cdot \sin(\alpha) + \frac{5}{7} \cdot \frac{M_r}{m \cdot r} = 0 \quad (32)$$

When the ball rolling on a spherical surface from a low level to a high level following differential equation can be written:

$$\frac{d^2x}{dt^2} + \frac{5}{7} \cdot g \cdot \sin(\alpha) + \frac{5}{7} \cdot \frac{M_r}{m \cdot r} = 0 \quad (33)$$

In the equations (32) and (33) following parameters are included:

- x is the distance from the ball to the center of spherical surface;
- t is the time, in seconds;
- g is the gravity acceleration (9.81 m/s²);
- α is the angular position of the ball, depending of the time in the motion of the ball;
- m is the mass of the ball;
- r is the radius of the ball;
- M_r is the rolling friction torque in the contact between ball and surface.

The rolling friction torque M_r can be expressed as a linear function of the rolling friction coefficient between the ball and surface μ_r :

$$M_r = \mu_r \cdot m \cdot g \cdot r \quad (34)$$

For given dimensions of spherical surface and balls, the free oscillations of the ball are depending of the friction losses in ball – surface contact. For various conditions, the free oscillations of the ball

was registered with a video camera and the amplitudes of the ball oscillations was experimental determined. Differential equations (32) and (33) was numerical solved according to the experimental values of the ball amplitudes during oscillating. The numerical values for a medium rolling friction coefficient have been obtained.

The experiments was made with three steel balls with diameters of 9.5 mm, 3 mm and 1 mm. The glass spherical surface was made with the radius of spherical surface $R = 75$ mm and with the diameter of 70 mm. The maximum value of the angle α was 18 degree.

To determine the influence of the capillary effects the experiments was made in atmosphere with the relative humidity of 60% to 80% at two temperatures of the glass surface: 10 °C and 70°C. The most important results was presented in the figure 10.

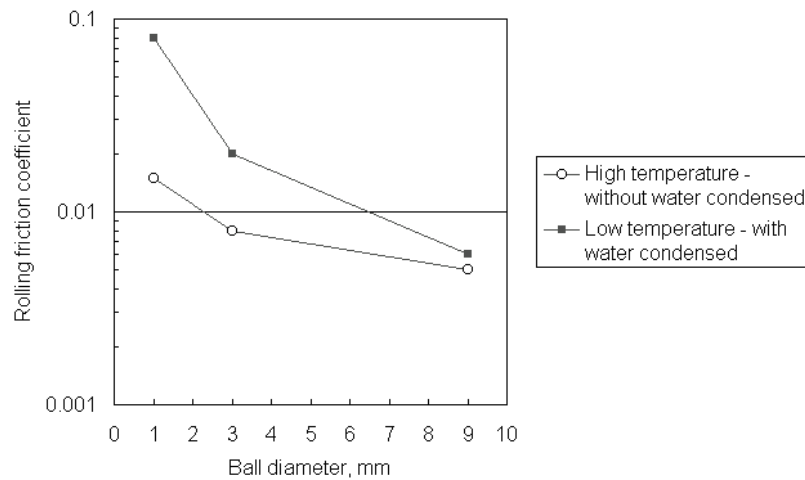


Figure 10 Experimental rolling friction coefficient

Some important conclusions can be obtained by the experiment:

- it is possible to estimate rolling friction coefficient by free oscillation of a ball over a spherical surface;
- for a macro ball (with diameter of 9.5 mm) rolling friction coefficient between steel and glass is about 0.005 – 0.006, with small increasing caused by capillary effects;
- for a quasi micro ball (with diameter of 1 mm) capillary effects lead to an important increasing of the rolling friction coefficient, from 0.015 at high temperature without water condensed on surfaces to 0.08 at low temperature with water condensed on surfaces;

5. CONCLUSIONS

A complex analytical model to evaluate the friction coefficient in a micro linear ball bearing has been developed. The model includes following losses sources: pivoting motion between ball-race, elastic resistance in rolling of the ball over the races, hydrodynamic effects in ball-races and the

losses in the ball-ball contacts. Both adhesion between balls and the races and capillary effects in ball - ball contacts was considered.

To validate the model, the numerical results was performed for a micro linear ball bearing studied experimental by [1]. The numerical results are in the same range of values with experimental results.

When the balls are not in direct contact one with other, the friction coefficient have values between 0.007 to 0.012 and can be increase with increasing the load to 0.03. For this condition the most important source of losses is the ball-race pivoting motion.

When the balls are in direct contacts one with other, the friction coefficient increases with an order of magnitude. So, values for friction coefficient between 0.25 and 0.55 was obtained. The friction losses between the ball-ball contacts are dominate.

To evidence the major role of capillary effects on the rolling friction a simple experiment based on the free oscillations of a steel ball over a spherical glass surface was developed. For a ball with diameter of 1 mm the capillary effects lead to an important increasing of rolling friction coefficient, from 0.015 to 0.08. For a ball with diameter of 9.5 mm, the capillary effects have small influence on rolling friction coefficient.

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Authors:

Professor Dumitru N. Olaru, Ass. Professor Gheorghe Prisacaru
Technical University “Gh. Asachi” of Iasi, Bld. D. Mangeron 61-63
700050, Iasi, Romania

Phone/Fax: +4-0232-232337

E-mail: dolaru@mec.tuiasi.ro gprisaca@mec.tuiasi.ro

Professor Peter Lorenz

Hochschule für Technik und Wirtschaft des Saarlandes – HTW, Goebenstrasse 40
D-66117 Saarbrücken, Germany

Phone +49 681 58 67 – 262, Fax: +49 681 58 67 – 122

E-mail: lorenz@htw-saarland.de